NUMERICAL SIMULATION OF THE AIR FLOW AROUND THE ARRAYS OF SOLAR COLLECTORS

by

Vukman V. BAKIĆ [*]***, Goran S. ŽIVKOVIĆ, and Milada L. PEZO***

Laboratory for Thermal Engineering and Energy, Vinča Institute of Nuclear Sciences, University of Belgrade, Belgrade, Serbia

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This paper deals with the numerical simulation of air flow around the arrays of flat plate collectors and determination of the flow field, which should provide a basis for estimating a convective heat losses, a parameter which influences their working characteristics. Heat losses are the result of the reflection on the glass, conductive losses at the collector's absorber plate, radiation of the absorber plate and convective losses on the glass. Wind velocity in the vicinity of the absorber plate depends on its position in the arrays of collectors. Results obtained in the numerical simulation of flow around collectors were used as boundary conditions in modeling of thermal-hydraulic processes inside the solar collector. A method for coupling thermal-hydraulic processes inside the collector with heat transfer from plate to tube bundle was developed, in order to find out the distribution of the temperature of the absorber plate and the efficiency of the flat plate collectors. Analyses of flow around arrays of collectors are preformed with RNG k-ε model. Three values for free-stream velocity were analysed, i. e. 1 m/s, 5 m/s, and 10 m/s, as well as two values for the angle between the ground and the collector (20° and 40°). Heat transfer coefficient was determined from the theory of boundary layer. Heat transfer inside the plate cavity was analyzed assuming constant intensity of radiation.

Key words: *solar collector, heat losses, turbulent flow, CFD*

Introduction

With the trend of increasing use of solar energy, which has attracted more attention in recent years, analyses and testing of solar collectors characteristics are gaining on importance. One of its most important parameter is heat loss, *i. e.* the amount of solar energy that reaches the plate collector, but fails to be handed over to the warm water. These losses are the result of reflection on the glass, conductive losses at the collector's absorber plate, radiation of the absorber plate, and convective losses on the glass. The first of the two above-mentioned losses depend exclusively of the characteristics of the collector. Loss by radiation depends on the temperature of the plate, as well as the ambient temperature. It is a function of total heat flux of solar radiation and total heat loss. Convective losses are primarily a function of air velocity (wind strength) over the glass panels. This paper analyzes the influence of the

* Corresponding author; e-mail: bakicv@vinca.rs

wind around collectors on performance of the collector *i. e.* temperature of the collector plate and efficiency of the collector. It is common in classical engineering approach to assume constant wind speed throughout the panel, which is not the case in practice. In addition, the velocity of air on the collector depends on its position in the array of collectors. Finally, in literature it is easy to find a lot of models of radiation and natural convection, which could be used for the calculation of heat loss. It is necessary to use, instead of conventional engineering heat loss calculation, commercial CFD software, which could accurately include all these phenomena. At the beginning, it is necessary to model flow of air around the collector plates, which gives the profiles of air velocity on a collector plates, depending on the average air velocity and position of the collector in array. The resulting profiles represent the boundary conditions for the next model, which includes thermal-hydraulic processes in the collector. Here, however, appears a common problem in using commercial CFD software, *i. e.* a commercial software is not generally capable to include appropriately two independent fluid flow of two different fluids (in the collector there is air in the cavity between absorber and collector plate, and hot water that flows through the heat exchanger that is fixed to the bottom of the collector plate). One possibility to overcome this problem and to quickly and efficiently determine temperature profile on the collector plate, as well as the efficiency of the solar collector is presented in this paper. It was performed by coupling the thermal-hydraulic processes in the collector's cavity with the processes of heat transfer from the plate to the arrays of heat exchangers tubes and by using certain iterative procedure described below.

Mathematical model

Mathematical model was partialy based on numerical and experimental researches found in literature. Naeeni *et al.* [1] performed two-dimensional numerical simulation of turbulent flow around a parabolic collector taking into account the effects of variation of collector angle of attack, wind velocity and its distribution with respect to height from the ground. They observed recirculation regions on the leeward and forward sides of the collector, and both pressure field around the collector and total force on the collector are determined for each condition. The effect of absorber tube on the flow field was found negligible, while the effect of the gap between the two sections of parabola at midsection and the gap between the collector and ground were found considerable on both flow field and pressure distribution around the collector. Chung *et al.* [2] have done experimental investigation to examine the aerodynamic characteristics of solar collector models (residential and large-scale solar water heaters). Measurements of mean longitudinal and spanwise surface pressure were also performed in this experiments. They shown that the presence of a water storage tank (or a horizontal cylinder) tends to reduce the suction force on the upper surface within the first half of the tilt flat panel (solar collector). Stronger negative longitudinal differential mean pressure is observed for the test case of a tilt flat panel only, which corresponds to strong wind load. The inverted U-shape of mean spanwise pressure distributions is also noted. Fan *et al.* [3] theoretically and experimentally investigated the flow and temperature distribution in a solar collector panel with an absorber consisting of horizontally inclined strips. Fluid flow and heat transfer in the collector panel were studied by means of CFD calculations. Further, experimental investigations of a 12.5 m^2 solar collector panel with 16 parallel connected horizontal fins were carried out. The flow distribution through the absorber was evaluated by means of temperature measurements on the backside of the absorber tubes.

In this paper, two-dimensional stationary 2-D RNG k-ε model, whose basic equations were given in the work of Yakhot *et al.* [4], was used for the analysis of flow around solar collectors. Yakhot *et al.* have proposed a variation of the k-ε (RNG) model. It is improved standard k-ε model. RNG turbulence model was used for the modeling of flow with sudden changes of streamline curvature, flow separation, and recirculation. It has application in the models which deal with the problem of wind flow, [5-7].

Flow field around the collector was analyzed for the air velocity in the undisturbed flow of 1 m/s, 5 m/s, and 10 m/s and the collector inclination angle of 20° and 40° to the horizontal. For the heat transfer coefficient calculation, the most appropriate term is the expression proposed in [8]:

$$h = 5.74V^{0.8}L^{-0.2} \tag{1}$$

where V is the air (wind) velocity around the collector, provided in m/s. Dimensions of the flow domain used for the modeling was 20 × 10 m. Array of three panels was analyzed, where the distance between them was 4 m and 5.3 m, which correspond to the minimum distance at which there is no mutual shading of collectors of the Sun's rays at 13 hours. The height of the flow field was 10 m, sufficient to fully cover the local variations of air velocity. For the modeling of the air flow field a numerical mash of 80,000 triangular control volumes was adopted. Inlet velocity profile U_{in} was given with the expression [6]:

$$U_{\text{in}} = 0.68U_{\infty}y^{0.17} \tag{2}$$

where U_{∞} is the undisturbed wind velocity, and y – the height in meters. As can be seen from (2), at 10 m height the input velocity becomes equal to U_{∞}, which justifies the chosen height for the modeled flow field.

Heat transfer in the collector was analyzed for the intensity of solar radiation of I = = 1040 W/m^2 and ambient temperature of 27 °C, corresponding values for a typical meteorological year in Belgrade on July 15, at 13 hours. The losses of reflection on the glass and the conductivity of the absorber plate were taken in account by using coefficient of 0.85, with which the intensity of the incident solar radiation should be multiplied.

Heat transfer in the solar collector was analyzed in two parts:

- heat transfer in the space between plates of the collector. The dimensions of the considered areas were 2.2 × 2 × 0.04 m. Thermal-hydraulic analysis of this region allows the determination of the plate temperature. For the numerical modeling of this area a rectangular structured numerical mesh of 4,000 control volumes was used. For the modeling of the heat transfer process in the collector (between two plates) a model for laminar flow was adopted, taking into account a small distance between the two plates and very low air velocity in the collector due to natural convection. For the modeling of radiation from the collector plate to the environment a Rosseland's model was proposed;
- heat transfer from the collector plate to the heat exchanges tubes, through which the hot water flows and is warmed up. In this paper a simple geometry of the heat exchanger was assumed, which consists of input collector placed across the solar collector (whose task was to redistribute water among the tubes along the entire solar collector), and the output collector, which is identical to the input collector and is used to collect water before it is sent to the heat storage tank. Input and output collectors were not covered by thermodynamic analysis. The uniform mass flow rate distribution among the tubes of heat exchanger was also assumed.

The resistance to heat flow to the fluid results from the bond and the tube-to-fluid resistance:

$$k = \frac{1}{h_\mathrm{i}\pi D_\mathrm{i}} + \frac{1}{C_\mathrm{b}} \tag{3}$$

where D_i = 10 mm is the inner diameter tube, h_i = 300 W/m^2 K – the heat transfer coefficient on the water, and C_b = 30-35 W/mK, [9], – the bond conductance between the plates and tubes with water.

The heat flux (per unit length) from the plate to water is given by:

$$q_{in} = \frac{1}{k}[T_\mathrm{p}(x) - T_\mathrm{w}(x)] \tag{4}$$

where T_p is the plate temperature, T_w – the water temperature in tubes, and x-co-ordinate is the co-ordinate along the collector. By setting the energy balance equation for the flow through n = 10 tubes of length l = 2.2 m at constant total mass flow of water $\dot{m}$ = 0.03 kg/s (Re = 687 laminar flow) linear differential equation for the temperature of water in the tubes was obtained, as a function of the temperature of the plate:

$$\dot{m}c_\mathrm{pw}\mathrm{d}T_\mathrm{f} = n\frac{T_\mathrm{p} - T_\mathrm{f}}{k}\mathrm{d}x \tag{5}$$

$$\frac{\mathrm{d}T_w}{\mathrm{d}x} + aT_\mathrm{w} = aT_\mathrm{p} \tag{6}$$

Here, $a = n/(k\dot{m}c_\mathrm{pw})$. The solution of this linear differential equation is given by expression (7), where A represents a constant of integration:

$$T_\mathrm{w} = \mathrm{e}^{-ax}(A + a\int \mathrm{e}^{ax}T_\mathrm{p}\mathrm{d}x) \tag{7}$$

Based on the previous experience it was assumed that the plate temperature T_p can be very well approximated by the polynomial of the fourth degree:

$$T_\mathrm{p} = a_4x^4 + a_3x^3 + a_2x^2 + a_1x + a_0 \tag{8}$$

After substituting the expression (8) in (7), integration and determination of the constant A, one obtains expression for the temperature of water in the tubes T_w (9), specific heat flux from the plate to the water q_in (10) and the total amount of heat that is transferred from the plate to water (11):

$$\begin{aligned} T_\mathrm{w} = {} & T_\mathrm{win}\mathrm{e}^{-ax} - a\left(\frac{a_0}{a} - \frac{a_1}{a^2} + \frac{2a_2}{a^3} - \frac{6a_3}{a^4} + \frac{24a_4}{a^5}\right)\mathrm{e}^{-ax} + \\ & + a\left[\frac{a_4}{a}y^4 + \left(\frac{a_3}{a} - \frac{4a_4}{a^2}\right)x^3 + \left(\frac{a_2}{a} - \frac{3a_3}{a^2} + \frac{12a_4}{a^3}\right)x^2 + \right. \\ & \left. + \left(\frac{a_1}{a} - \frac{2a_2}{a^2} + \frac{6a_3}{a^3} - \frac{24a_4}{a^4}\right)x + \left(\frac{a_0}{a} - \frac{a_1}{a^2} + \frac{2a_2}{a^3} - \frac{6a_3}{a^4} + \frac{24a_4}{a^5}\right)\right] \end{aligned} \tag{9}$$

$$q_{in} = \frac{1}{k}(T_\mathrm{p} - T_\mathrm{w}) = \frac{1}{k}\Bigg\{ -T_\mathrm{win}\mathrm{e}^{-ax} + a\left(\frac{a_0}{a} - \frac{a_1}{a^2} + \frac{2a_2}{a^3} - \frac{6a_3}{a^4} + \frac{24a_4}{a^5}\right)\mathrm{e}^{-ax} + a\left[\frac{4a_4}{a^2}x^3 + \left(\frac{3a_3}{a^2} - \frac{12a_4}{a^3}\right)x^2 + \left(\frac{2a_2}{a^2} - \frac{6a_3}{a^3} + \frac{24a_4}{a^4}\right)x + \left(\frac{a_1}{a^2} - \frac{2a_2}{a^3} + \frac{6a_3}{a^4} - \frac{24a_4}{a^5}\right)\right]\Bigg\} \quad (10)$$

$$\int_0^l q_{in}\mathrm{d}x = \frac{1}{k}\Bigg\{\left(\frac{a_0}{a} - \frac{a_1}{a^2} + \frac{2a_2}{a^3} - \frac{6a_3}{a^4} + \frac{24a_4}{a^5} - \frac{T_\mathrm{win}}{a}\right)(1 - \mathrm{e}^{al}) + a\left[\frac{a_4}{a^2}l^4 + \left(\frac{a_3}{a^2} - \frac{4a^4}{a^3}\right)l^3 + \left(\frac{a_2}{a^2} - \frac{3a_3}{a^3} + \frac{12a_4}{a^4}\right)l^2 + \left(\frac{a_1}{a^2} - \frac{2a_2}{a^3} + \frac{6a_3}{a^4} - \frac{24a_4}{a^5}\right)\right]\Bigg\} \quad (11)$$

Numerical procedure was carried out in several steps:

– on the basis of the assumed temperature of the plate distribution T_p(x) calculations of T_w(x), q_in, and the total amount of heat transferred from the plate to the water in the tubes are performed, eqs. (7), (8), and (9);
– boundary conditions for the flow field between the collector plates are defined (heat transfer coefficient from glass to the surrounding air, heat flux on the plate, adiabatic isolated outside walls), and numerical simulation are carried out, to determine the temperature of plate T_p(x). A relevant heat flux on the plate for the calculated flow field is heat flux of solar radiation that falls on the plate minus the convective heat flux from the plate to the water;
– the assumed temperature T_p in the first step is replaced with the temperature obtained in a second step (using appropriate under-relaxation factor, if necessary), and the iterative process of (1) through (3) is repeated. Iterative procedure is repeated until the profiles of temperature T_p and the total amount of heat transferred from the plate to the water from the tubes in two successive iterations coincide within the desired accuracy.

Results of numerical simulation

Velocity field around the collector for the air velocity of 5 m/s and for the collector inclination angle to the ground of 20° and 40° are shown in fig. 1. In all modeled cases, the maximum of velocity was on the first collector, which was to be expected. Velocity on the

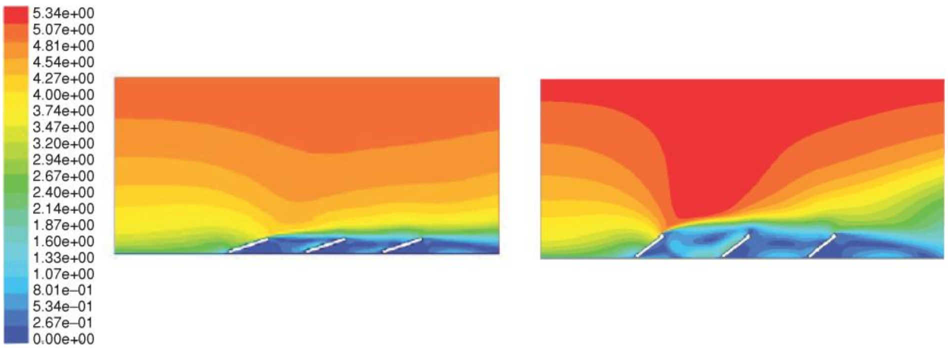

Figure 1. Velocity field around collectors for wind velocity of 5 m/s for collector angles of 20° (left) and 40° (right); (color image see on our web site)

second and third collector was much smaller, with a little difference between them. Flow field is similar in all considered cases. The values of heat transfer coefficient for each collector was calculated after determining the velocity profile close to the boundary layer on the upper surface of the collector, by using the expression (1).

Using an iterative procedure described in the "Mathematical model" section, changes in temperature between the glass and the collector plates were obtained. Obtained flow velocity in the cavity was very small, and practically can be ignored. This is also an expected result, because of the small height of the flow field. Temperature difference between glass and absorber plate on the first collector is lower than the second and third collector at all considered air velocities. The difference in temperatures was between 20 °C and 40 °C from one case to another.

Figure 2 shows changes in air temperature between the glass panels on all three panels at an angle of 20° and air velocity of 5 m/s.

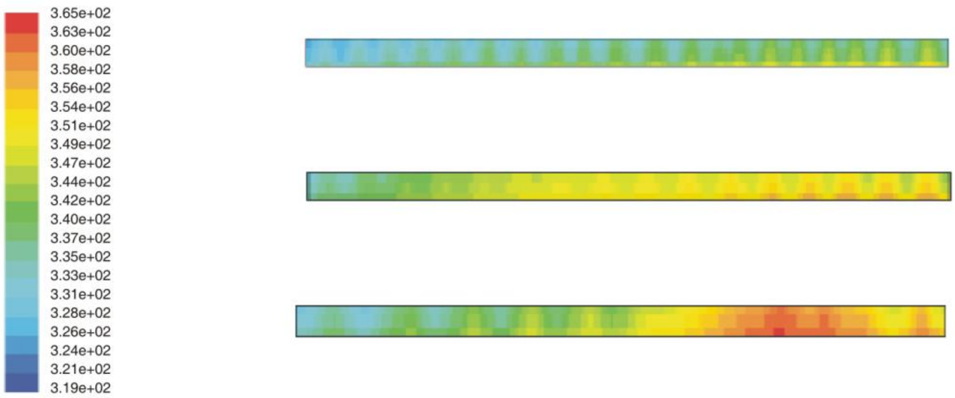

Figure 2. Changes in air temperature between the glass and the receiving plate collectors for collector inclination angle of 20° and the air velocity of 5 m/s for the first, second, and third collector; (color image see on our web site)

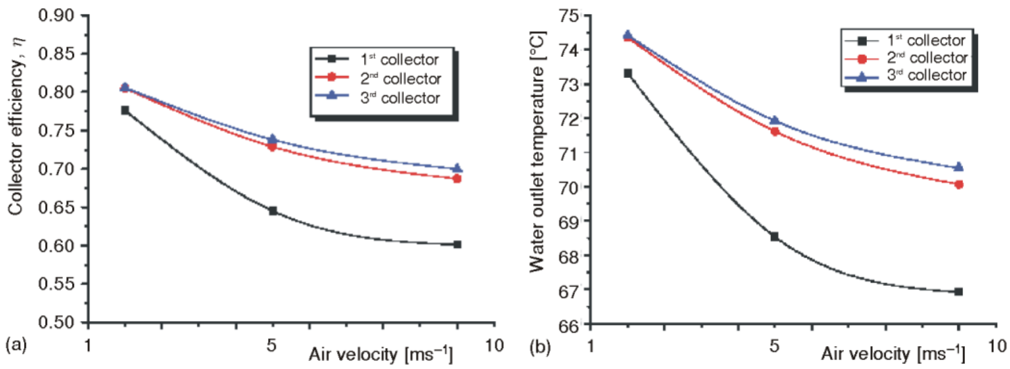

Figure 3. Collector efficiency (left) and water outlet temperature (right) for air velocity of 1 m/s, 5 m/s, and 10 m/s and the collector inclination angle of 20°

Change of the outlet water temperature and the changing of the collector efficiency η defined as the ratio of heat flux of water and loss of sunlight are shown in fig. 3. The calculations were carried out for inlet water temperature of 45 °C, which is an average outlet temperature of the hot water tank, when users are connected to the system (results shown in lines).

It is clear from fig. 3 that the influence of air velocity is the most evident on the first collector. The collector efficiency drops from 81% to 61% with the increase of the air velocity from 0 to 10 m/s, while the outlet water temperature decreases from 74.5 °C to about 67 °C. The influence of the air velocity on the second and third collector is much smaller, which leads to the conclusion that the change of the collector's performance in other rows of collectors is the same with the change obtained for the second row. The temperature of water at the inlet also significantly affects the calculated η.

From fig. 4 it is evident that the influence of air velocity is mostly evident on the first collector. The collector efficiency with the increase of the air velocity drops from 81% to 60% with the increase of the air velocity from 0 to 10 m/s, while the output water temperature decreases from 75 °C to about 66 °C. The influence of the air velocity on the second and third collector for the collector inclination angle of 40° is significantly different than for the collector inclination angle of 20°. Influence of the distance between the collector and the collector inclination angle significantly affects the performance of flat-plate collectors.

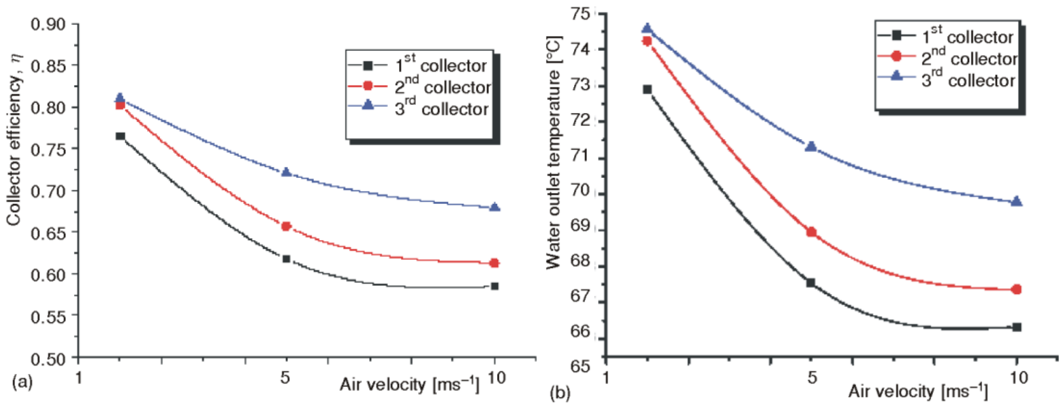

Figure 4. Collector efficiency (left) and water outlet temperature (right) for air velocity of 1 m/s, 5 m/s, and 10 m/s and collector inclination angle of 40°

Conclusions

This paper analyzes effects of air velocity around collectors on solar collector parameters for different inclination angles of 20° and 40°. Heat loss, *i. e.* the amount of solar energy that reaches the plate collector, but fails to be handed over to the warm water is the most important parameter and it is observed in this paper. These losses are the result of reflection on the glass, conductive losses at the collector's absorber plate, radiation of the absorber plate and convective losses on the glass. The first of the two above-mentioned losses depend exclusively of the characteristics of the collector. Loss by radiation depends on the temperature panels (and ambient temperature) as a function of total heat flux of solar radiation and total heat loss. Convective losses are primarily a function of air velocity (wind strength) over the glass panels.

Stationary 2-D RNG k-ε model was used for numerical simulation. It was necessary to model flow of air around the collector plates. As an results of this modeling profiles of air velocity on a collector plates were obtained. They depend on the average air velocity and the position of collector in array. The resulting profiles represent the boundary conditions for next model, which includes thermal-hydraulic processes in the collector.

Numerical experiments were performed for array of three collectors. They showed that the effect of air velocity on the collector's performance in the first row is significant and that a drop in efficiency of these collectors is 20% for the air velocity of 10 m/s. Numerical experiment also showed that the air temperature between the plates and glasses is more then 30 °C higher for collectors in the first row then for the others.

The increase of the collector inclination angle leads to the formation of different velocity field around the collector, which causes a significant impact on the performance of collectors.

For proper designing the array of collectors and in their use in obtaining hot water the wind velocity at a given location has to be taken into account.

Nomenclature

A – constant of integration
a – $n/k\dot{m}c_{pw}$, [m^{-1}]
C_b – bond conductance between the plates and tubes with water, [Wm^{-1}K^{-1}]
c_{pw} – specific heat of water for constant pressure, [Jkg^{-1}K^{-1}]
D_i – inner tube diameter, [m]
h – heat transfer coefficient, [Wm^{-2}K^{-1}]
h_i – heat transfer coefficient inside tubes, [Wm^{-2}K^{-1}]
I – intensity of solar radiation, [Wm^{-2}]
l – length, [m]
q_{in} – heat flux (per unit length) from plate to water, [Wm^{-1}]
T_p – plate temperature, [°C]
T_w – temperature of water in tubes, [°C]
U_{in} – inlet velocity profile, [ms^{-1}]
U_∞ – undisturbed wind velocity, [ms^{-1}]
V – air velocity, [ms^{-1}]
x – co-ordinate along the collector, [m]
y – height, [m]

Subscripts

i – inner
in – inlet
p – plate
w – wall
∞ – undisturbed

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